15th Australasian Fluid Mechanics Conference
The University of Sydney, Sydney, Australia
13-17 December 2004

A Numerical Study of Enhanced Micro-channel Cooling Using a Synthetic Jet Actuator

V. Timchenko, J. Reizes and E. Leonardi

School of Mechanical and Manufacturing Engineering
The University of New South Wales, Sydney 2052, Australia.

Abstract

The use of micro synthetic jets to disrupt a laminar flow in forced air convection micro-channels and improve a heat transfer to the cooling air has been studied numerically. The effectiveness of the proposed cooling strategy was evaluated by comparing heat transfer rates with and without the synthetic jet. Simulation results indicate that the location of the synthetic jet is important and that synthetic jets have the potential to significantly improve heat transfer from integrated circuits.

Introduction

The evolution of integrated electronic devices has led to substantial increases of heat generation rates and requires the development of innovative cooling techniques. Forced air convection in silicon micro-channels etched in the rear side of the silicon substrate has the potential to remove heat directly from the chip and has been developed as one of the strategies for cooling of integrated circuits after micro-channel heat sink concept was first introduced in Tuckerman and Pease [1] pioneering study. However, airflow in such conduits is laminar with the result that the heat transfer rate is quite low. In addition, if the upper surface of the micro-channels is a hot surface, the heat transfer rate may be further reduced by the appearance of separations near the inlets on the upper surfaces of the channels.

A synthetic jet [2], so-called because the jet is synthesized from the working fluid without introducing a fluid from another source, can be used for disrupting the laminar flow in the channels as well as for interfering with the stagnant zones. In fact, the unsteady flow from the jet creates a quasi-turbulent flow, thereby possibly significantly increasing the rate of heat transfer. The synthetic jet is developed in the cooling fluid by a micro-pump actuator [3] which consists of an oscillating diaphragm in a cavity with a small orifice in the face opposite the diaphragm as may be seen in Figure 1. Under appropriate operating conditions this actuator provides a fluctuating flow away from the orifice into the cooling channel. The actuator thus has a zero nett mass flow, but a non-zero nett momentum transfer over an entire cycle of the diaphragm oscillation.

Utturkar *et al.* [4] suggested the jet formation criterion $1/St = \mathrm{Re}/S^2 > K$, in which the constant K was said to be approximately 2 for two-dimensional jets and approximately 0.16 for axisymmetric jets. The non-dimensional parameters used in their analysis of synthetic jet actuators were the Reynolds number, $Re = \overline{V}_j d/\nu$, in which, $\overline{V}_j$, is the average orifice expulsion velocity, d is the orifice width, and ν is the kinematic viscosity, the Stokes number, $S = \sqrt{\omega\, d^2/\nu}$, in which $\omega = 2\pi f$ is the angular velocity and f is the frequency of the diaphragm motion and the Strouhal number, $St = \omega d/\overline{V}_j$.

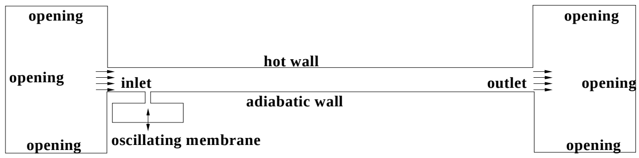


Figure 1. Sketch of computational domain.

An experimental study of periodic disturbances on the penetration and mixing of synthetic jets in cross-flow was performed by Eroglu and Breidenthal [5]. They observed that periodic forcing of the jet stream effects mixing by creating vortex loops whose strength and spacing are determined by the frequency of the forcing and the jet to cross-flow velocity ratio. The main parameter characterizing the jet in cross-flow has been found to be the jet-to-cross-flow momentum ratio [6].

In this paper we study numerically the feasibility of using synthetic jet actuators to enhance heat transfer rates in micro-channels. A two-dimensional micro-channel 200 μm high and 4.2 mm long is considered with a top surface hot and all the other walls adiabatic (Figure 1). The effectiveness of the proposed cooling procedure is evaluated by comparing the heat transfer rates with and without the synthetic jet.

Mathematical and numerical model

For micro sized devices the flow is dependent on the Knudsen number, Kn, which is defined as ratio of the mean free path to the characteristic geometry length ($Kn=\lambda/L$). In this study flows with Kn < 0.01 are modeled, so that the continuum approach using conventional conservation equations is still valid [7].

The flow generated by synthetic jet actuators has been simulated using a commercial package, CFX-5.6. Unsteady computations of compressible laminar flow have been performed for two-dimensional numerical formulations. The basic set of the unsteady conservation equations for laminar flow comprises the continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \tilde{V}) = 0\,, \qquad (1)$$

the Navier-Stokes equation

$$\frac{\partial \rho \tilde{V}}{\partial t} + \nabla \cdot \rho \tilde{V}\tilde{V} = -\nabla p + \nabla . \left(\mu \nabla \tilde{V} + \mu (\nabla \tilde{V})^T \right) + \rho \tilde{g} \qquad (2)$$

and the energy equation

$$\frac{\partial \rho h}{\partial t} + \nabla \cdot (\rho \tilde{V} h) - \nabla \cdot (k \nabla T) = 0\,, \qquad (3)$$

in which ρ, $\tilde{V}$, p, μ, h, T, k, and t denote density, velocity vector, pressure, viscosity, enthalpy, temperature, thermal conductivity and time respectively. The compressibility of the air was taken into account by treating it as an ideal gas with an equation of state given by

$$\rho = \frac{M(p + p_{ref})}{R_u T}\,, \qquad (4)$$

in which M is the molecular mass of the gas, R_u is the universal gas constant, T is the absolute temperature and p_{ref} is the absolute ambient pressure.

The displacement of the membrane Y_m was assumed to be a parabola which varies sinusoidally in time, viz,

$$Y_m = A(1 - (x/r)^2)\sin(2\pi f t) \qquad (5)$$

in which A and f are the amplitude and the frequency of oscillation respectively and r is the radius of the membrane.

A second order backward Euler differencing scheme was used for the transient term, whereas a second order upwind differencing scheme was used for the advection terms in the Navier Stokes equation. At each time step (equal to one hundredth of a cycle), the internal iterations were continued until the mass and momentum residuals had been reduced to 10^{-6}.

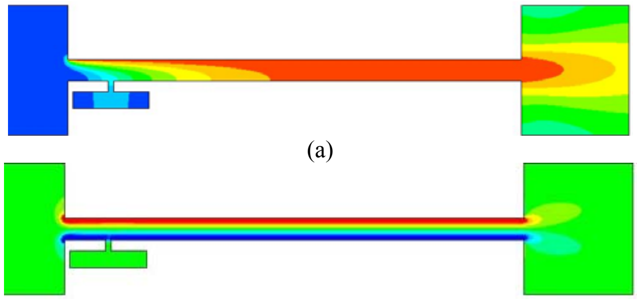
The number of grid points used in the orifice of the synthetic jet generator was 50 x 20 in the stream-wise and transverse directions respectively, so mesh size was kept to 2.5 μm in the orifice. Outside the orifice grid was gradually expanded with maximum mesh size equal to 5 μm. The total number of mesh points was equal to 164316.

To study the effect of synthetic jets on the enhancement of heat transfer in the micro channel we introduce the factor of heat transfer effectiveness F, defined as $F = \dot{Q}/[(1/2)\dot{m}U^2]$, in which $\dot{Q}$ is the average heat transfer rate from the upper wall, $\dot{m}$ is the mass flow rate, U is an average axial velocity in the channel. This factor represents a ratio of the heat transferred to the total flow kinetic energy.

Results and discussion

Two two-dimensional cases are presented here. In the first (Case 1) an orifice 50μm wide and 100μm long was placed 400μm downstream from the inlet. The width of the diaphragm was 0.7*mm* and the cavity depth was 250μm. As inflow temperature and velocity distributions affect the heat transfer in the channel, external domains 1*mm* long and 1.5*mm* high were included at either end of the channel (Figure 1). In Case 1, an average velocity in the channel of about 1 *m/s* was obtained with a pressure drop of 25*Pa*, the temperature was set to 20º C at the inlet of the (left) external domain. In Case 2, the pressure was increased to 250 *Pa* with the temperature remaining at 20º C at the inlet of the external domain. At the outlet (right) of the external domain, in both cases, the pressure was fixed to zero with p_{ref} equal to 100*kPa* and the temperature set to 30º C. The upper wall of the channel was isothermal at the 50º C.

To evaluate the effectiveness of the proposed cooling strategy the steady flow heat transfer was used as the standard. In this particular case we wanted to observe an ability of the synthetic jet to disrupt the laminar flow near the inlet of the channel. Thus we positioned it 1.2 *mm* from the inlet.



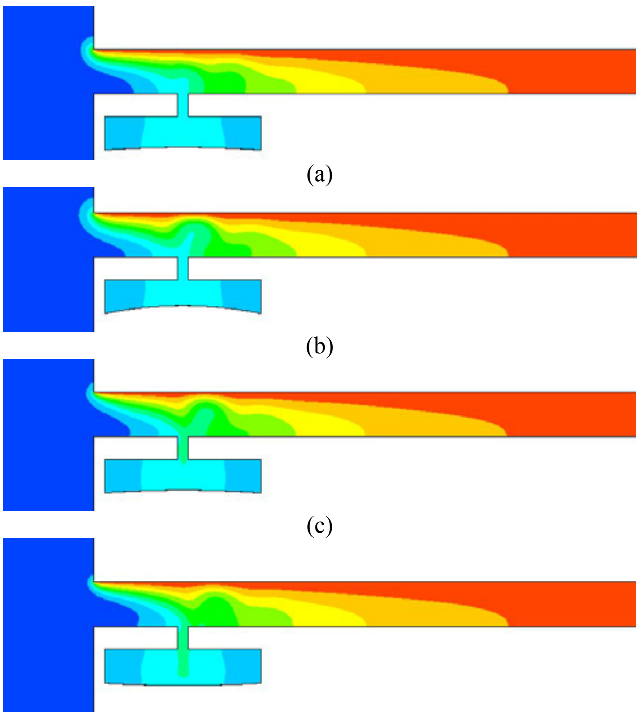
(a)

(b)

Figure 2. Temperature (a) and vorticity (b) contours in the case of steady flow (Case 1).

Figure 2 shows temperature and vorticity contours in the case of steady flow with the jet not active. Horizontal contours of vorticity correspond to a laminar, parabolic flow with no mixing occurring. The heat transfer rate from the hot upper wall in the case of the steady flow with mass flow rate, $\dot{m} = \rho A U$ equal to $0.24\text{x}10^{-3}$ *kg/sm* and $\dot{Q}$, the rate of heat transfer was $59.0\text{x}10^{-1}$ *W/m*. The resulting heat transfer effectiveness factor was F = 4600. It should be noted from the temperature contours in Figure 2 that the average fluid temperature reaches the very close value to that of the upper wall about 1.8*mm* from the inlet, so that there is little heat transfer between that point and the outlet.

Enlarged views of the temperature and vorticity contours at various instants of the cycle are shown in Figures 3-4. A jet was produced by the displacement of membrane with frequency 5*kHz* and the amplitude 23.6μm. During the expulsion stage a vortex is created near the orifice. It travels a short distance towards the upper wall of the channel thereby interfering with the boundary



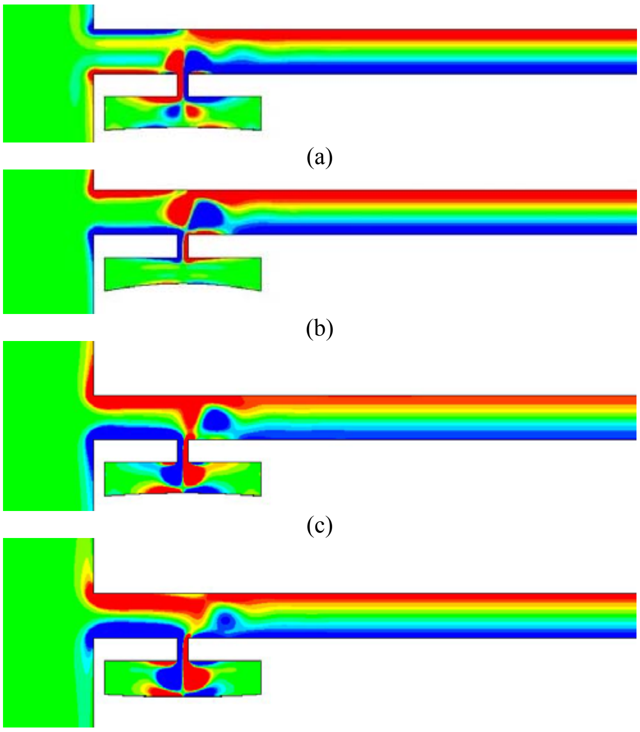
(a)

(b)

(c)

(d)

Figure 3. Temperature contours at the Maximum Expulsion (a), Minimum Volume (b), Maximum Ingestion (c) and Maximum Volume (d) stages (Case 1).



(a)

(b)

(c)

(d)

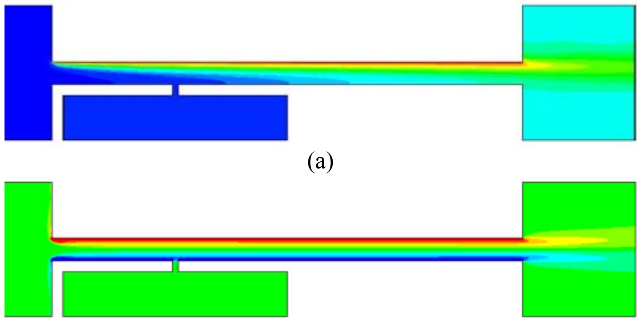
Figure 4. Vorticity contours at the Maximum Expulsion (a), Minimum Volume (b), Maximum Ingestion (c) and Maximum Volume (d) stages (Case 1).

layer in which heat transfer occurs. Unfortunately, the disturbance does not propagate very far downstream. In Case 1 the heat transfer rate cannot be increased very much above that in the steady state with the result that $\dot{Q}$ = 60.8x10^{-1} *W/m* when average mass flow rate becomes is 0.242x10^{-3} *kg/sm* and average downstream flow velocity *U*=1.022 *m/s* with F = 4813. Thus, for Case1 the heat transfer effectiveness factor increased by only about 5% when the membrane was activated.

By actuating with the chosen forcing membrane parameters a synthetic jet with *Re*=15 and *1/St*=2.8 was produced. It should be noted that when there is a cross-flow, the jet is not continuous despite the fact that in a still atmosphere a continuos synthetic jet with would have been produced [8]. The jet-to-cross-flow momentum ratio during expulsion stage defined as $C_{\exp} = d\overline{V}_j^{\,2} / DU^2$ where *D* is the height of the channel equal to 5.

Because in Case 1 substantial changes were generated by the jet on the inflow temperature and velocity distributions (see Figures 3-4), in Case 2, the orifice has been moved further downstream to a position 1.2 mm from the inlet of the channel. The pressure drop was increased to 250 *Pa* to obtain a higher mass flow rate. It may be seen from Figure 5, that in Case 2, the steady flow temperature distribution indicates that it should be possible to significantly increase the rate of heat transfer. In steady flow, the heat transfer rate from the hot upper wall was 30.6*W*/m, whilst the mass flow rate was 1.81 x 10^{-3}*kg/sm* and the average velocity in the channel was 8.06*m/s*. The resultant heat transfer effectiveness factor was 520.

In case 2, the width of the diaphragm was 1*mm* and the cavity depth was 400μm so as to keep jet-to-cross-flow momentum ratio



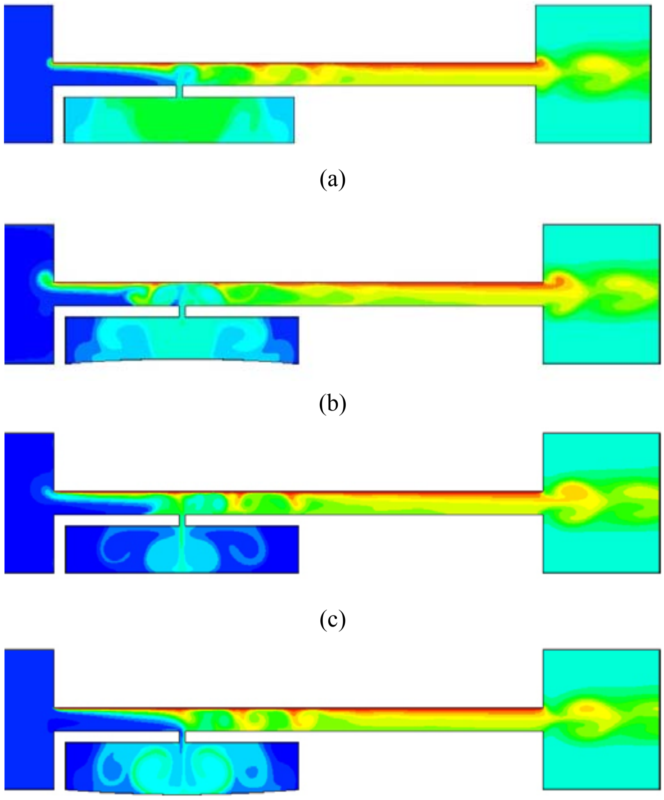
(a)

(b)

Figure 5. Temperature (a) and vorticity (b) contours in the case of steady flow (Case 2).

during expulsion stage equal to 5, the frequency was increased to 10*kHz* with the amplitude of 42μm. This led to a jet with *Re*= 125 and *1/St*=12. Temperature and vorticity contours at various stages in the cycle may be seen in Figures 6 and 7. It can be seen that moving the orifice further away from the inlet together with increasing of mass flow rate resulted in substantially deeper penetration into the channel main flow of the vortices created during the expulsion stage thereby significantly enhancing mixing.

As a consequence, the heat transfer rate from the hot upper wall averaged over period of oscillation is 36.05*W/m* with average mass flow rate equal to 1.68x 10^{-3}*kg/sm* and average downstream flow velocity of 7.09*m/s*. The heat transfer effectiveness factor is *F* = 853.Thus in the Case 2 the heat transfer effectiveness factor in the case of oscillating jet increased by approximately 64%.



(a)

(b)

(c)

(d)

Figure 6. Temperature contours at the Maximum Expulsion (a), Minimum Volume (b), Maximum Ingestion (c) and Maximum Volume (d) stages (Case 2).

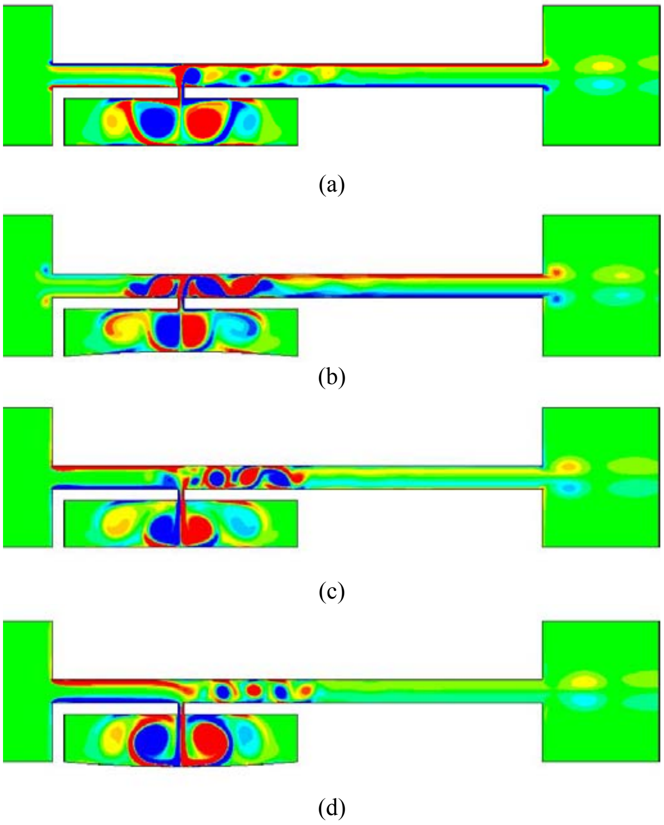


Figure 7. Vorticity contours at the Maximum Expulsion (a), Minimum Volume (b), Maximum Ingestion (c) and Maximum Volume (d) stages (Case 2).

Conclusion

In this preliminary study simulations have been performed to evaluate the feasibility of using micro synthetic jet actuator to disrupt a laminar flow and enhance heat transfer in micro-channels. The initial results indicate that a fully optimized synthetic jet has the potential to significantly improve heat transfer from an integrated circuit to the cooling fluid. A detailed parametric study will be performed in order to find the optimal parameters, which lead to the largest enhancement of heat transfer.

Acknowledgments

The authors would like to acknowledge with thanks the support of the Australian Research Council.

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